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Performance and control of domestic ground-source heat pumps in retrofit installations

P.J. Boait[a,*], D. Fan[a], A. Stafford[b]

[a] Institute of Energy and Sustainable Development, De Montfort University, Queens Building, The Gateway, Leicester LE1 9BH, UK
[b] Centre for the Built Environment, Leeds Metropolitan University, Queens Square Court, Leeds LS2 8AJ, UK

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ABSTRACT

Heat pumps are an essential technology for decarbonisation of domestic heating in the UK. This paper reports on the performance in use of a group of ground-source heat pumps, and in common with other UK studies finds that the seasonal performance is not as good as that reported in trials from continental Europe and that the system controls are unsatisfactory. Control improvements are investigated via a model of the dwelling and heat pump as a combined system, from which the thermal time constant of the building is identified as a critical factor that needs to be considered in retrofit projects incorporating heat pumps. The validity of the conventional practice (and advice from installers to users) of allowing heat pumps to run continuously is tested and bounded. Techniques for improving control are outlined and reasons for the poorer performance in the UK examined with the conclusion that heat pumps need to be better matched in capacity and control to the size and thermal characteristics of UK dwellings. Implementation of these findings by heat pump manufacturers and installers could promote a more rapid transition to renewable heat both in the UK and internationally wherever similar housing stocks and climates exist.

1. Introduction

The Energy White paper of 2007 [1] committed the UK government to providing support for low-carbon technologies, and encouraging energy saving in the domestic sector through better information, incentives and regulation. More recent strategy documents aim to reduce carbon emissions from this sector by almost 30%, with a projection that by 2020 up to 7 million homes will have received eco-upgrades, including measures such as solid wall insulation and heat pumps [2]. Social housing providers, including local authorities, have been tasked with providing leadership in such upgrade programmes, for example via CESP (Community Energy Saving Programme), in partnership with community groups and energy companies [3].

Heat pumps are also an eligible technology for the proposed Renewable Heat Incentive programme, which was expected to come on-stream in April 2011. There is now some doubt that this scheme will be ratified in its original form by the present Coalition Government, but nevertheless it is still anticipated that some form of support for low-carbon heat generation will be offered. Therefore it is confidently to be expected that heat pumps will play an increasingly important part in the UK domestic sector as a retrofit low-carbon technology [4]. The present work has been undertaken as part of the Carbon, Control and Comfort (CCC) project. CCC is a cross-disciplinary project aimed at developing techniques for reducing carbon emissions while maintaining desired comfort levels, using action research and user-centred design approaches to assess the effects of both improved control technology and social issues surrounding control systems.

Harrogate Borough Council in North Yorkshire currently has around a hundred social housing properties with retro-fitted ground-source heat pumps installed. A sub-group of ten of these properties has been intensively monitored for heat pump performance and associated energy usages. We present here a summary of the results from this monitoring and the user experience which show that there is scope for improvement in the controls of these devices when used in this type of retrofit application. To investigate possible control regimes, data collected from these heat pumps has been used to develop and validate a simple model representing as a single system both the operation and coefficient of performance (CoP) of these heat pumps and the thermal properties of the building in which they are installed. Using the model, we identify opportunities and methods for improved control and

* Corresponding author. Tel.: +44 124 251 1278; fax: +44 116 257 7981.
E-mail address: p.boait@dmu.ac.uk (P.J. Boait).

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Nomenclature

a	slope of linear regression function for μ
b	intercept of linear regression function for μ
C	thermal capacity of building (kW h/°C)
CoP	coefficient of performance of heat pump
E_{in}	electrical energy input to heat pump (kWh)
K	radiator circulation temperature control constant
L	heat loss rate of building (W/°C)
Q_{out}	thermal energy output from heat pump (kWh)
R	radiator heat transfer coefficient (W/°C)
t_0, t_s, t_{min}	times for start, finish, and minimum temperature of setback
T_s	room temperature set point (°C)
T_e	external ambient temperature (°C)
T_L	temperature lift of heat pump (°C)
T_g	ground loop output temperature (°C)
T_r	radiator return temperature (°C)
T_{min}	minimum room temperature during setback (°C)
W	heat input power to building (kW)
W_h, W_a, W_o	heat input from heat pump, appliances, and occupants (kW)

Greek letters

μ	multiplying factor to derive practical CoP from Carnot ratio
τ	thermal time constant of building

also constraints arising from the long thermal time constant of well insulated buildings with high thermal mass.

2. Description of dwellings monitored

2.1. Building, heat pump and occupant characteristics

The monitored dwellings are all rurally located one- or two-bedroom bungalows, built between 1967 and 1980, and let by Harrogate Borough Council as social housing for the elderly. Fig. 1 shows a typical example. All have been upgraded with cavity-wall insulation, double-glazing and additional loft insulation. The heat pumps were installed during the winter of 2007–2008, almost two years before monitoring commenced.

The systems studied are all ground-source heat pumps (IVT Greenline HT Plus C6) with bore-hole collectors, connected to a conventional wet central heating system with radiators oversized by around 30% by comparison with conventional UK practice for gas fired boiler installations. Space-heating and domestic hot water (DHW) are both supplied by the heat pump, with the assistance of a 3 or 6 kW inbuilt electric resistance heater, which is brought incrementally online only as a supplement where necessary, typically during the weekly hot water pasteurisation cycle.

The tenants (either single occupants or couples) were in most cases given only limited information about the operation of the heat pump, and generally advised not to alter controls, but to contact the council or their agents if adjustments were required. They were, however, advised that opening windows would be likely to affect the efficiency of heat pump operation adversely.

2.2. Monitoring methods and instrumentation

A schematic illustration of the monitoring instrumentation is shown in Fig. 2. *E* represents Elster AC100 kW h meters (pulse output resolution 1 pulse per Wh), located to measure total electricity consumption of the heat pump, and also the contributions of the electric cassette (resistance heater), the hot-side pump and the ground loop pump. The energy used by the compressor and associated control system and displays is obtained by subtraction. *H* represents Supercal 539 Compact Heat Meters (pulse output resolution 1 pulse per 100 W) located such as to measure total heat pump output, and output to the central heating circuit. Output to DHW is obtained by subtraction. Points marked *T* show the locations of K-type surface mount thermocouples. *F* is a cold water flow meter (pulse output resolution 1 pulse per litre), located on the replacement cold water flow into the DHW storage tank, and thus yielding the volume of DHW used.

In addition to the meters shown on the diagram, temperature and humidity was monitored in four locations within the dwelling, CO_2 in one location, and external conditions were monitored via a weather station comprising a Vaisala WXT520 weather transmitter and Kipp & Sonnen CMP3 pyranometer with integral logger. Data from all meters and sensors was transmitted wirelessly via Eltek Gen II transmitters to loggers (Eltek Squirrel RX250) at 10 min intervals. This data was then downloaded remotely at regular intervals via a GSM modem.

A key parameter measured and modelled throughout this paper is room temperature. This is air temperature as sensed by a microprocessor-based device (resolution 0.1 °C, accuracy 0.4 °C over the range encountered), located towards the centre of the building where it is not directly affected by convection currents or draughts. It is recognised that this does not fully characterise temperature as perceived by an occupant, but it is similar to the conventional low cost techniques employed by heating control systems that are the main theme of the paper and therefore air temperature is the focus of the analysis.

3. Initial results from the intensive study

3.1. Heat pump performance

During the study period the heat pumps operated in the continuous mode recommended by the manufacturers and installers. This employs a primary control mechanism that aims to maintain a nominal room temperature set point T_s by measuring external ambient temperature T_e and regulating the return temperature T_r from the radiators so that $(T_r - T_s)/(T_s - T_e)$ is a constant K. This constant (approximately the ratio between the radiator heat transfer coefficient R and the heat loss coefficient L of the house) is a parameter that is initially set by the installer (known as the operating slope or curve) and can be varied by the user. This control method assumes that these two coefficients are known and fixed for the dwelling and that there are no other variable sources of heat; the limitations arising from this assumption are discussed in Section 5. There is however a summer disconnection facility allowing the heat pump to produce DHW only when the external temperature exceeds a pre-set value (usually 16 °C). Stored DHW is kept at a pre-set temperature except during the weekly pasteurisation cycle when the temperature in the tank is raised to >60 °C for a period of at least half an hour.

During the winter period of the heating season the heat pumps were able to maintain constant indoor temperatures with little or no use of the additional electric cassette, even during periods of very cold weather. In summer, with no output from the heat pump to space-heating, indoor temperatures could exceed target values due to solar gain. Fig. 3 shows the month by month CoP of all the systems, defined as:

$$\mathrm{CoP} = \frac{Q_{out}}{E_{in}} \tag{1}$$

over a given time period (in this case one calendar month), where Q is the total heat output from the heat pump, and E is the electrical

Fig. 1. An example of the houses monitored.

energy consumed. Note that in this case E_{in} excludes the energy use of the hot-side distribution pump. Some data points have been excluded where the system was known to be malfunctioning during part of a particular month.

There is a wide spread of CoP values due to different settings and other factors, but generally all systems follow a similar trend of increasing CoP values during the spring season when space-heating temperature lifts are lower, followed by a reduction in the summer season when the heat pump is producing mostly DHW. It is expected that the spring season maximum will be replicated during autumn. These results are consistent with those obtained by other UK studies, notably that by the Energy Savings Trust [5] who monitored 47 GSHP installations and obtained a range of CoPs from 1.2 to 3.2 with a median value of 2.2. There is a consistent pattern of

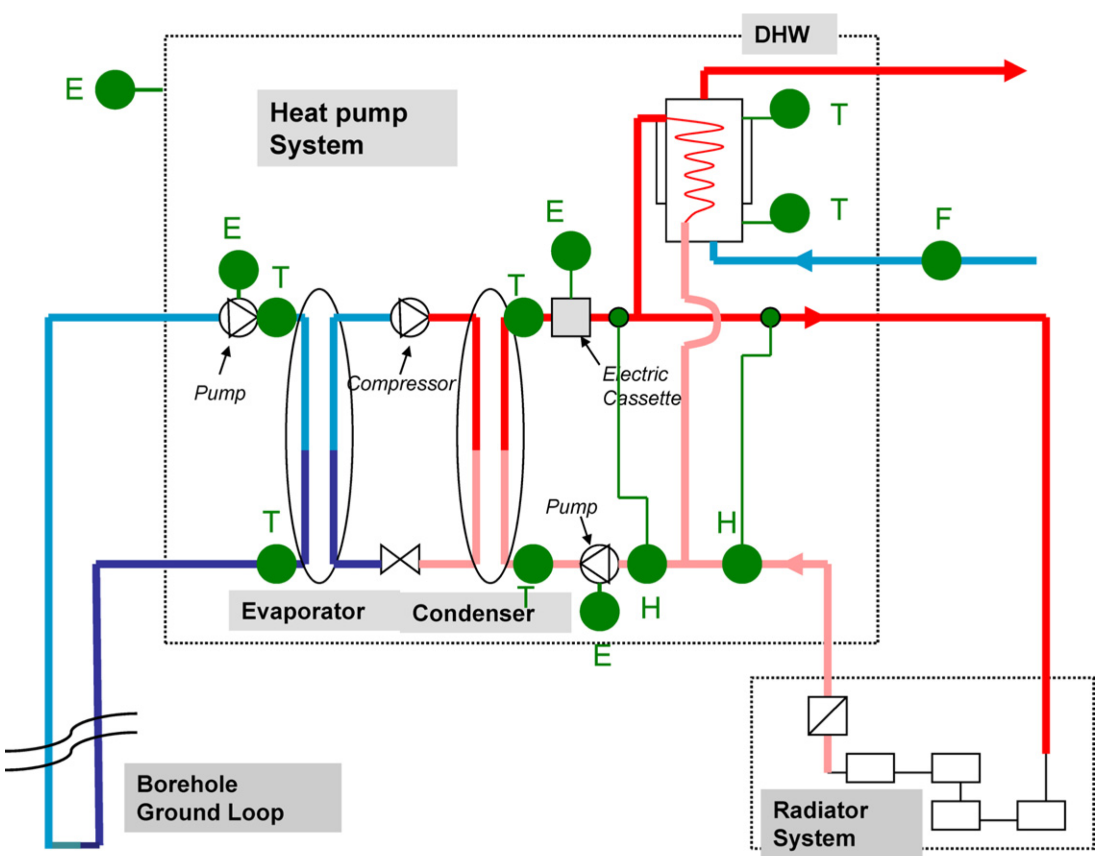

Fig. 2. Schematic diagram of monitoring instrumentation.

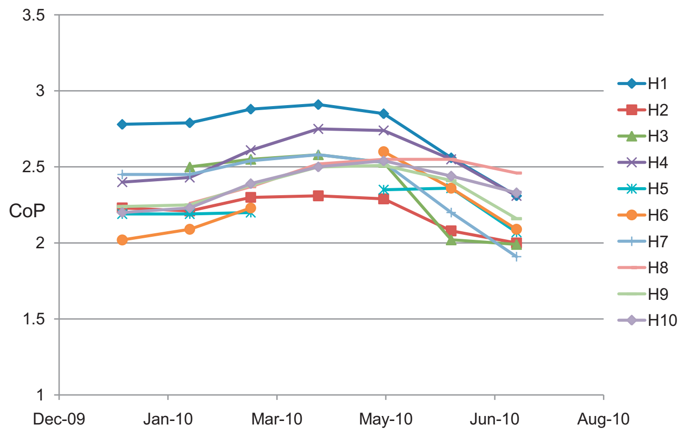

Fig. 3. Monthly system CoP values (January to July 2010).

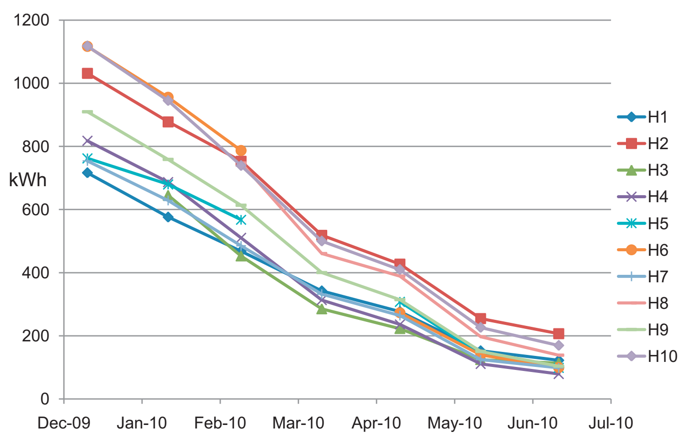

Fig. 4. Monthly electricity use by heat pumps (January to July 2010).

relatively poor results from UK installations compared to European experience such as that reported by the Fraunhofer Institute for Solar Energy Systems [6] who obtained average CoPs of 3.3 for existing dwellings and up to 4.6 for new build, while Marcic [7] reports an average of 3.16. Possible reasons for this significant difference are discussed in Section 6. The electricity consumption of each heat pump system (this time including the hot-side distribution pump) is summarised in Fig. 4.

The data is subject to the same exclusions as in Fig. 3. Although the monthly electricity usage is affected to some extent by factors such as length of month and number of weekly pasteurisation cycles included in that month, the clear downward trend shows that external temperatures are the primary driver. The data for January 2010 are particularly significant since this was an exceptionally cold winter for the UK but the average consumption of 900 kW h and CoP of 2.4 imply a continuous heat output of 2.9 kW which is only 58% of the heat pump capacity of 5 kW.

3.2. User experience

Occupants generally reported good satisfaction with the levels of thermal comfort provided by the heat pump systems, though some felt that expected cost savings had not been realised.[1] Two common areas of dissatisfaction were a perceived lack of control (possibly associated with low confidence in operating the user interface) and a feeling that the high night-time temperatures provided by the system are undesirable. The latter point may be the result of actual feelings of discomfort (many people preferring cooler temperatures at night) or may be associated with the belief that night-time operation is wasteful, since the concept that switching the heat pump off during sleeping hours would actually result in higher energy usage remains somewhat counter-intuitive. In at least one case, the problem is known to be 'resolved' by opening windows at night to counter excessive heating.

Table 1
Comparison between room temperature set point and actual temperature.

House #	Room temperature set point	Average measured living room temperature (December)
1	21	23.9
4	24	19.5
5	20	20.1
6	22	22.3
7	21	20.4
9	23.5	20.8

While the heat pump operation is primarily controlled by specifying a pre-set target radiator return temperature as a function of external temperature, there is also a facility for allowing a room thermostat setting to influence heat output. However because the return temperature control ratio *K* is difficult to set accurately and there is an exponential lag in the response of room temperature (discussed further in Section 5) these two control mechanisms may interact unpredictably. This is illustrated by Table 1, where the average measured temperature in living rooms (the site of the room thermostat) for the month of December 2009, was poorly correlated with the room temperature thermostat setting in some cases, and well correlated in others. This difficulty in achieving comfortable and efficient control of heat pumps in UK dwellings has been noted by other studies, particularly [5], which notes that "control systems were generally too complicated for the householders to understand" and "some householders found it difficult to control the ambient room temperature", leading to one of its key conclusions that "Heating controls for heat pump installations have to be comprehensively reviewed."

3.3. Control requirements

A first requirement for any attempt to improve heating controls is that it should improve what Baker and Standeven [8] called "adaptive opportunity" – the perception by occupants that they can adjust the heating if they need to. This of itself enhances their comfort and willingness to tolerate variations in temperature. The practical implication is that the control interface should be very easy to understand and use. This will also avoid the widely reported misuse of controls – e.g. by Shipworth et al. [9]. Clearly a simple, reliable and accurate method of setting room temperature is required, so that users can express their varying needs and preferences. Ideally, for comfort and efficiency the room temperature set point should be adjusted automatically to reflect longer term variations in external ambient temperature [10,11].

Operating a heating system continuously throughout the year with weather compensation of the circulating water temperature and the summer cut-off mechanism described in Section 3.1, but no simple means to reflect variations in occupancy or desired room temperature, does not seem likely to be an optimum control mode.

[1] Occupants opinions were collected in a survey by Dr Debra Lilley of Loughborough University – see acknowledgements. Her findings will be covered more extensively in a future publication.

There must be some periods when the dwelling is unoccupied or the occupants require a lower room temperature, during which a useful cessation or reduction in heat pump output could be performed. Such an interval is referred to here and in heat pump manufacturer's documentation as a setback. The heat pump equipment does recognise this possibility by accepting a "night setback" input from an external control device, but its use is discouraged by installers. Clearly therefore it is preferable that a control system should itself be able to determine and operate setback periods that save energy and/or improve comfort.

4. Building and heat pump model

4.1. System modelling method

Any form of closed-loop control (i.e. control which adjusts an input to a system in order to achieve a desired output such as a specific room temperature set point) needs to include a representation of the transfer function by which a change in the input parameter translates into a change in the output parameter. For closed loop control of room temperature the transfer function must represent the building and the occupants (to the extent that their metabolism and use of appliances provides a heat input) as well as the heat pump. Because of the difficulty in constructing this transfer function for a very wide range of installations, heat pump manufacturers typically do not attempt it but rely on the form of open loop control described in Section 3.

However, since we are concerned specifically with retrofit installations in small UK dwellings, a proper characterisation of the building, occupants, and heat pump as a single system is a reasonable ambition. Since the transfer function must be implemented within the control sub-system, it needs to be as simple as possible consistent with adequate accuracy. For the building, a first order (single node) representation of the form:

$$(T_r - T_e) = \tau \frac{dT_r}{dt} \tag{2}$$

is proposed where T_r and T_e are room and external temperatures and τ is the thermal time constant of the building (= thermal capacity C kW h/°C divided by heat loss rate L kW/°C). The brick construction and good insulation of the houses in this study put τ at about 50–60 h which unavoidably constrains the responsiveness of any control mechanism.

To model the heat pump a simple method is to take the Carnot ratio and modify it with a multiplier μ that represents the mechanical and other losses so that:

$$\text{Practical CoP} = \frac{\mu(T_L + T_g + T_e + 273)}{T_L} \tag{3}$$

T_L is the temperature lift between the heat pump output to the radiator circuit and the ground loop output temperature and T_g is the temperature difference (which should typically be fairly small) between the external ambient temperature and the ground loop exit temperature. Regression analysis of the performance of the heat pumps monitored in this study provided a satisfactory dependence of μ on heat output power W of the form:

$$\mu = aW + b \tag{4}$$

This approach to heat pump modelling has been employed by other authors, for example Magraner et al. [12]. Its effect is that the reduction in CoP due to increased T_L that arises with increasing heat output from a heat pump is partly offset by an increasing μ, which captures the proportionate reduction in mechanical and electrical losses as the heat pump is working harder.

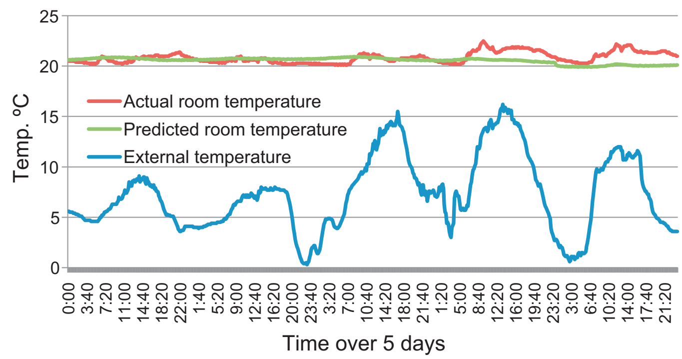

Fig. 5. Validation of first order building model.

4.2. Model validation

The accuracy of the simple building model given in (2) was tested by comparing actual room temperatures measured over 5 days from one of the monitored dwellings with those predicted by the model. Input data to the model were the starting room temperature, actual external temperatures, and actual measured heat power inputs for the period from the heat pump (W_h) and appliance use (W_a), plus assumed metabolic output (W_o = 100 W) from a single occupant. The room temperature was predicted at 10 min time steps from the heat balance:

$$C\partial T_r = (W_h + W_a + W_o)\partial t - L(T_r - T_e)\partial t \tag{5}$$

Fig. 5 shows an illustrative result. It can be seen that the prediction tracks the measured temperature closely despite wide variations in external temperature, with slightly greater deviance in the last two days when there was significant solar gain. The high room temperature on the last day also illustrates a situation where the open loop control of room temperature was not optimum – the heat pump was running during that afternoon and evening driven by the fall in external temperature even though room temperature was well above set point. The other discrepancies between prediction and measurement are likely to be the result of uneven distribution of heat inputs within the building, for example when internal doors are closed, and short term variations in heat loss caused by opening windows or external doors.

The heat pump model was tested by comparing the actual electrical load consumed by heat pumps from the trial over 24 h with that predicted by the model from the measured heat output and temperature lift. A single pair of regression parameters (a = 0.2, b = 0.033) was used for all the heat pumps to compute μ. The results are shown in Fig. 6 – the three groupings of data points reflect tests on data from January, April, and July. From these tests it was concluded that these simple models would be fit for use by a control system that must predict the outcome of control decisions over time scales related to τ and minimise electricity consumption. The use of a single node representation of the building limits such a control system to small dwellings where measurement of air temperature at a single point provides adequate regulation. The heat pump model also requires a static radiator heat transfer coefficient in order to predict temperature lift – this clearly precludes thermostatic radiator valves (which are deprecated for heat pump installations but sometimes used) and will be problematic if users operate radiator valves manually. Similarly variations in heat loss rate will disrupt the building model, but as these are typically caused by opening or closing windows or doors their impact is well understood and intended by occupants.

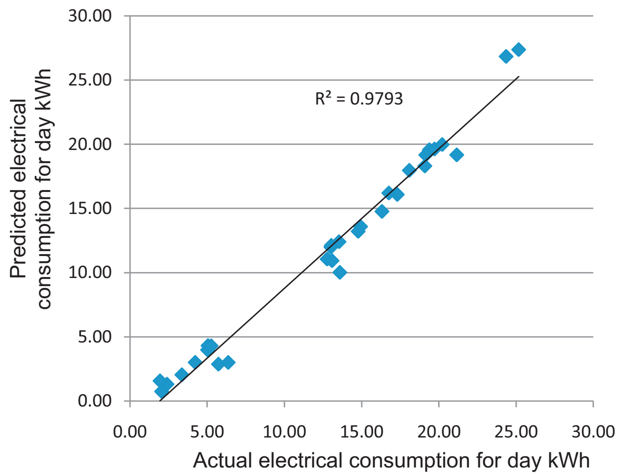

Fig. 6. Validation of heat pump model.

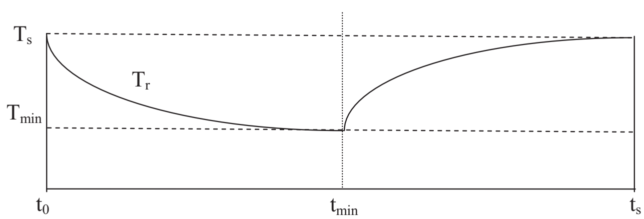

Fig. 7. Setback parameters for analysis.

4.3. Analysis of setback

With this system model it is possible to test whether the conventional practice of running heat pumps continuously is correct, or alternatively it is possible for an optimum duration and temperature profile of a setback period to be calculated so that energy consumption is reduced or at least held constant and comfort is improved. Because of the simple nature of the model, an analytic solution of useful generality is possible. Fig. 7 illustrates the scenario under consideration.

The objective is to compare the energy used to maintain a room temperature set point T_s throughout a given time interval t_s with that required if a setback is performed where heating is ceased at t_0 and restarted at t_{min} such that with a constant heat input W from the heat pump T_s is recovered at t_s. In the setback case the room temperature follows the profile marked as T_r. To simplify presentation of the analysis, T_s, T_r, T_{min} and the ground loop output temperature T_g are all defined as being relative to the external ambient temperature T_e (i.e. room temperature in Kelvin would be $T_r + T_e + 273$).[2] The high τ typical of well insulated brick-built houses, and the relatively low heat output of heat pumps, means that a minimum acceptable overnight temperature of say 16 °C will not be reached even in winter before it is time to recommence heating so that the daytime set point is recovered. Thus there is no need for a lower minimum temperature set point to be defined within the setback period.

To calculate the electrical energy used by the heat pump in each scenario we need to calculate T_L which is given by:

$$T_L = \frac{W}{R} + T_r - T_g \tag{6}$$

[2] For clarity of presentation the approximate conversion from Celsius to Kelvin is used here, but for calculations using this system model the value of 273.15 is preferred.

where R is the heat transfer coefficient of the radiators. Then substituting (4) and (6) in (3) gives:

$$\mathrm{CoP} = \frac{(aW + b)(W/R + T_r + T_e + 273)}{W/R + T_r - T_g} \tag{7}$$

For the scenario where the room temperature is held constant, T_s can be substituted for T_r to obtain the electrical energy E consumed giving:

$$E = \frac{Wt_s}{\mathrm{CoP}} = \frac{Wt_s(W/R + T_s - T_g)}{(aW + b)(W/R + T_s + T_e + 273)} \tag{8}$$

If T_g and T_e are assumed to be constant over t_s this is readily evaluated.

To determine the energy used in the setback case we have to perform the integration:

$$E = \int_0^{t_s - t_{min}} \frac{W}{\mathrm{CoP}} dt \tag{9}$$

The first step is to find t_{min}. This can be obtained beginning with the requirement that T_s is recovered at t_s, so:

$$T_s = T_s e^{-t_{min}/\tau} + (T_m - T_s e^{-t_{min}/\tau})(1 - e^{-(t_s - t_{min})/\tau}) \tag{10}$$

where T_m is the maximum differential between room temperature and external temperature that can be supported by W i.e. W/L. Multiplying out (10) and simplifying gives:

$$T_s = T_m - T_m e^{-(t_s - t_{min})/\tau} + T_s e^{-t_s/\tau} \tag{11}$$

Dividing throughout by T_m and rearranging gives:

$$e^{-(t_s - t_{min})/\tau} = 1 + \frac{T_s}{T_m}(e^{-(t_s/\tau)} - 1) \tag{12}$$

Multiplying throughout by $e^{t_s/\tau}$ gives:

$$e^{t_{min}/\tau} = e^{t_s/\tau} - \frac{T_s}{T_m} e^{t_s/\tau} + \frac{T_s}{T_m} \tag{13}$$

Finally extracting t_{min} and rearranging gives:

$$t_{min} = \tau \ln\left[\frac{T_s}{T_m} + e^{t_s/\tau}\left(1 - \frac{T_s}{T_m}\right)\right] \quad \text{and} \quad T_{min} = T_s e^{-t_{min}/\tau} \tag{14}$$

The integration can now be performed by substituting (4), (6) and (3) in (9):

$$E = \int_0^{t_s - t_{min}} \frac{W(W/R + T_r - T_g)}{(aW + b)(W/R + T_r + T_e + 273)} dt \tag{15}$$

To simplify the integration, we can take the factor $(W/R + T_r + T_e + 273)$ as a constant T_K, since its numeric value is about 320, so the small change of a few °C in T_r over the period will not affect it by more than about 2%, and also represent W/R as T_R – the temperature difference between the radiators and the room. Taking constant factors out of the integration and putting T_r in terms of t gives:

$$E = \frac{W}{(aW + b)T_K} \int_0^{t_s - t_{min}} T_R - T_g + T_{min} + (T_m - T_{min})(1 - e^{-t/\tau}) dt \tag{16}$$

Then putting $T_m - T_{min} = T_D$ and integrating gives:

$$E = \frac{W}{(aW + b)T_K} \left[(T_R - T_g + T_{min} + T_D)t + T_D \tau e^{-t/\tau}\right]_0^{t_s - t_{min}} \tag{17}$$

Subtracting the evaluation of (8) from (17) indicates whether the setback costs or saves energy.

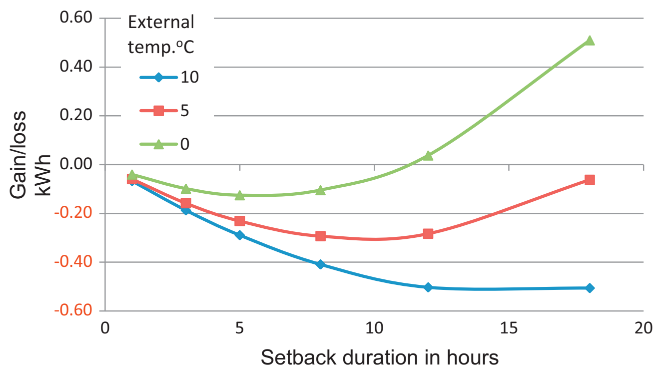

Fig. 8. Energy balance for setbacks in house A.

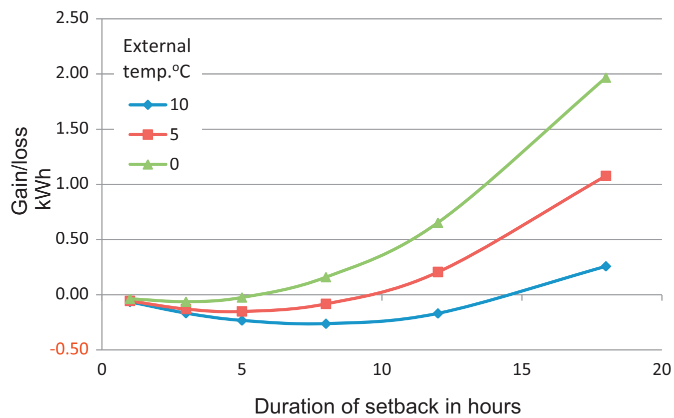

Fig. 9. Energy balance for setbacks in house B.

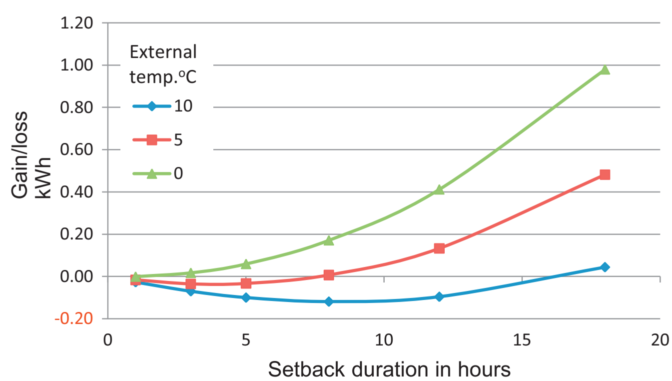

Fig. 10. Energy balance for setbacks in house C.

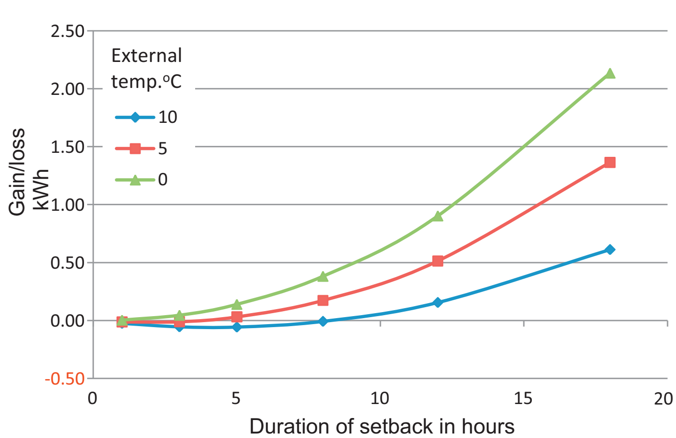

Fig. 11. Energy balance for setbacks in house D.

Table 2
Thermal parameters for modelled house-heat pump combinations.

House	A	B	C	D
Building heat loss rate (W/°C)	185	185	185	185
Building thermal capacity (kW h/°C)	10	5	10	5
Radiator heat transfer coefficient (W/°C)	120	120	240	240
Heat pump output (kW) for Te = 0 °C	4.44	4.44	4.44	4.44
Heat pump output (kW) for Te = 5 °C	3.52	3.52	3.52	3.52
Heat pump output (kW) for Te = 10 °C	2.59	2.59	2.59	2.59

4.4. Modelling results

Figs. 8–11 show the results from applying the method described in Section 4.3 to determine the gain or loss of energy arising from setbacks of increasing duration to four house and heat pump combinations (A–D) that are similar to those studied, but with varying thermal parameters listed in Table 2 to draw out the practical sensitivities and implications of the analysis. In all cases an accurate setting for K is assumed so that the heat pump output W is appropriate for T_e, R, and L. It is important to note that the length of time heating is actually off (the time to t_{min}) is less than half of the setback duration t_s for efficient levels of W. An 18 h setback can therefore provide a comfortable result if initiated at say 11pm when occupants retire to bed and t_{min} occurs at 7am when the occupants get up, with a T_{min} about 1 °C less than T_s. Room temperature will then rise gently during the day which is often compatible with perception of comfort – for example Hong et al. [13] in a Warm Front study show a difference of about 1 °C between the mean morning and evening temperatures at which Comfort Vote determined using ISO 7730 methods is neutral (i.e. zero).

House A (Fig. 8) has a high but not exceptional τ of 54 h. The energy balance is plotted for three levels of external temperature during the setback, and it can be seen that the balance is only positive for longer setbacks and an external temperature of 0 °C, even though the heat pump power was kept as low as possible during the recovery phase of each setback consistent with practical operation of setback as a control mechanism. This shows that the default recommendation from installers to users to run 24 h is reasonably efficient most of the time for houses with a high τ and conventional radiators.

Fig. 9 shows the results for house B, which is the same as A except that the thermal capacity is half that of A giving a τ of 27 h. It can be seen that operating a daily setback is a much more practical proposition, with energy savings from setbacks above 10 h for the lower levels of outside temperature. House C is the same as A, but with radiators having twice the heat transfer coefficient (e.g. by installing fan assisted convectors). The results are similar to house B, but with lower maximum gains and losses. Finally house D has both a lower τ and better radiators, and is amenable to setbacks of more or less any duration with negligible scope for losses and significant scope for energy savings. The monitored dwellings fall within the parameter space delineated by A, B, and C.

5. Implications for heat pump control

The first priority for improved control must be more accurate setting of K. The evidence from the monitored homes is that it tends to be set too high by installers, who err on the side of ensuring warmth for the occupants, resulting in a higher radiator temperature and lower CoP than would otherwise be the case. For a new build home it is a reasonable expectation that the architects should calculate R and L and apply a correction for appliance and metabolic heat inputs to arrive at a good estimate of K. In the case of a retrofit project this is unlikely to be practical unless a total refurbishment of the building is being performed. Preferably the heat pump control system should calculate R and L in operation using its own sensor measurements and set K automatically.

A problem for determination and operation of a fixed K in small well insulated dwellings is that appliance and metabolic heat inputs can be a substantial and variable proportion of the total heat requirement. In the 5 day period in April shown in Fig. 4 the daily heat demand varied in the range 20–30 kW h, while appliance and lighting inputs varied between 1.6 kW h and 3.8 kW h and the occupants body heat probably contributed about 2–4 kW h. Since occupancy is typically correlated with appliance use, these fluctuating inputs reduce the effectiveness of the radiator temperature control loop, although Fig. 5 shows that for these homes the resulting excursions in room temperature are nicely constrained by the high τ to about 1 °C until solar gain takes effect in the last two days.

The analysis in Section 4 indicates that although operating a heat pump continuously is a reasonable default recommendation, it will often result in missed opportunities for improved comfort and/or energy savings. A control system should therefore be able to calculate when a setback is likely to be efficient given the trend of external ambient temperatures at the time, and then apply it depending on occupancy as indicated by the user's heating time settings as entered manually into a conventional time programmer or detected automatically using the techniques described in [14]. This will require the control system to fully implement the method described in Section 4.

Room temperature should ideally be stabilised and varied using conventional closed-loop control which incorporates the first order building model as an exponential lag in the transfer function as described for example by Webb [15]. This would be beneficial in ensuring that the output from the heat pump is correctly modulated in response to variations in casual heat sources. However the difficulty for the occupants is that the perceptible responsiveness of their heating will always be constrained by τ so any attempt to achieve a rapid rise in temperature in a high-τ home will be penalised with a poor CoP, while a rapid fall in temperature can only be achieved by opening windows to effectively reduce τ by increasing L. Possibly the best approach for a control system is that it should combine the provision of clear information that accustoms users to this slow response time with the application of controlled setbacks of varying duration to shape the daily temperature profile and maximum temperature in accordance with their inputs. Development of a control system with these properties is the subject of further work under the present programme.

6. Discussion

The benefits in heat pump efficiency from radiators or underfloor heating with a high heat transfer coefficient are well understood by installation contractors so, in the case of the dwellings studied, radiators 30% oversized by comparison with gas boiler practice were installed. However the implications of high thermal mass on heat pump performance are less widely recognised. Many of the retrofit projects currently in progress in the UK are applying external insulation as a method of driving down heat losses, particularly for older buildings without cavity walls. When combined with improved air tightness the result will be a thermal time constant of several days. The resulting constraints on controllability need to be understood and tolerated by the occupants, and mitigated as far as possible by improved controls as discussed in Section 5, if the benefits of the investment in terms of reduced carbon emissions are to be fully realised. Also, where the dimensions and structure of the building permit it, insulation methods that do not increase the apparent thermal mass should be preferred where heating is by heat pump. For a household with variable occupancy (such as a working family with school age children) a lower τ combined with good radiators has the potential to make a real difference to their energy consumption as illustrated in Fig. 11. Where a high τ is unavoidable, perhaps as a consequence of exceptionally good insulation, then high performance radiators become even more important to maintain controllability.

Turning to the question of why heat pump performance in the UK is on average worse than in continental Europe, the first factor that is evident from this and other studies is the size of dwelling and corresponding heat load. The average floor area of the retrofit properties examined in [6] was 190 m^2 whereas the houses that form the subject of this study are about 60–80 m^2. Because the heat pumps currently available in the UK are designed for these larger homes the minimum capacity offered is 5 kW, which is much more than necessary to meet the winter load of a small well insulated home needing 100 W/°C. Consequently heat pumps in the UK are operating more lightly loaded and Eq. (4) dictates that if other parameters are normalised then mechanical losses will be proportionately higher, as will parasitic electrical loads such as circulation pumps, resulting in a lower average CoP. The higher proportion of heat pump output devoted to sanitary water heating will have the same effect. No doubt the smaller size of UK homes also leads to smaller radiators (since there is less internal wall area) and to a correspondingly worse CoP. A final factor is likely to be the much wider use of timber framed construction in northern continental Europe which will provide a low τ and the benefits identified above.

Three conclusions are therefore offered from this study. Firstly improved heat pump controls can and should be designed using the methods discussed that cope with the thermal properties of UK homes after retrofit improvements. Secondly, architects should be aware of the synergy between heat pumps and a low thermal time constant. Finally, heat pump manufacturers should be encouraged to produce units with smaller capacity more suited to the future UK market. In view of the greatly increased penetration of GSHP installations expected in the future, these findings are of considerable significance in the UK, and internationally where similar climatic conditions and housing stocks are found. Given widespread acceptance by the industry, they should lead to more acceptable designs on a much shorter timescale than if market inertia is allowed to set in, and thus permit more rapid and certain transition to renewable heat wherever the technology is appropriate. A major manufacturer of domestic GSHPs is being engaged directly to ensure that these conclusions have maximum impact.

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